1952

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Massachusetts Institute of Technology

http://hdl.handle.net/10945/14113

THE EFFECT OF SURFACE TEMPERATURE ON
THE STABILITY OF THE BOUNDARY LAYER

JOSEPH D. EVANS
WILBUR M. MORRISON
CHARLES E. SLONIM

THE EFFECT OF SURFACE TEMPERATURE
ON THE STABILITY OF THE BOUNDARY LAYER

by

ABSTRACT

Title of Thesis: The Effect of Surface Temperature on the Stability of the Boundary Layer.

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Submitted to the Department of Naval Architecture and Marine Engineering on 16 May 1952 in partial fulfillment of the requirements for the degree of Naval Engineer.

Based on the results of experimentation with gas flow, a theory has been proposed that for the case of water, increasing the temperature of a surface moving relative to the water should tend to stabilize the laminar boundary layer of the flow past the surface.

The object of this investigation was to determine if the laminar boundary layer can be stabilized by heating the surface of a vessel in water, and to gain experimental evidence as to the practical applicability of such a method of boundary layer control to reduce the frictional resistance of small submerged vessels.

The method employed to obtain the necessary information consisted of using an electrically heated, copper model hull which was towed in the conventional manner in the M.I.T. towing tank. It was felt that a significant effect on the stabilization of the laminar boundary layer would be apparent in the delay of the occurence of transi-

tion from laminar to turbulent flow.

Difficulties in providing sufficiently large quantities of power to the model to give a relatively high surface temperature, and difficulties in determining the exact nature of the flow in the unheated condition were encountered.

It is felt that the results obtained from heating the surface of the model can be accounted for by a pure viscosity effect within the limits of precision obtainable. No definite conclusions, therefore, as to the effect of heating on the stability of the laminar boundary layer can be reached. It is recommended that investigation of the effect be continued utilizing methods of experimentation wherein the characteristics of the flow can be determined and large quantities of power for heating can be readily introduced.

Thesis Supervisors: Ascher H. Shapiro
Professor of Mechanical Engineering

Martin A. Abkowitz
Assistant Professor of
Naval Architecture

Cambridge, Massachusetts
16 May 1952

Professor Leicester F. Hamilton
Assistant Secretary of the Faculty
Massachusetts Institute of Technology
Cambridge, Massachusetts

Dear Sir:

In accordance with the requirements for the degree of Naval Engineer, we submit herewith a thesis entitled, "The Effect of Surface Temperature on the Stability of the Boundary Layer."

Respectfully,

ACKNOWLEDGEMENT

The Authors wish to express their appreciation to Professors A. H. Shapiro and M. A. Abkowitz for their advice and assistance. It was Professor Shapiro's initial proposal for an investigation that resulted in this thesis. The authors also wish to express their appreciation to the personnel of the Boston Naval Shipyard, Professor J. N. Addoms, Professor W. H. McAdams, Professor H. C. Hottel, and Professor F. E. Vinal without whose cooperation and assistance this thesis could not have been undertaken.

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SYMBOLS:

C_f	Frictional- resistance coefficient
C_r	Residual-resistance coefficient
C_t	Total-resistance coefficient
R_e	Reynolds number
R_t	Total resistance
T_{as}	Average surface temperature
T_w	Water temperature
T_b	Effective boundary-layer temperature (see Appendix B)
L	Waterline length
S	Wetted surface area
ρ	Mass density of water
ν	Kinematic viscosity
V	Model Speed

I. INTRODUCTION

The magnitude of the frictional resistance of a vessel in water depends on whether the boundary layer is laminar or turbulent. The frictional resistance is very much greater when the boundary layer is turbulent. Therefore a delay in the transition from laminar to turbulent flow might result in a significant reduction in the resistance of vessels, particularly submerged vessels whose resistance is basically frictional resistance. Boundary layer control may be most readily applied to small craft such as torpedoes, where exhaust gases may be so routed as to heat the exterior surface of the vessel.

Experimentation [2, 3, 5, 6, 7, 8] and theoretical calculations [1, 4] have shown that in the case of gas flow, heating the surface of a vessel advances the point of transition and cooling delays the point of transition. The viscosity of air increases with an increase in temperature, while the viscosity of water increases with a decrease in temperature. It is possible, therefore, that the heating of a surface in water may delay the transition from laminar to turbulent flow.

The object of this investigation was to determine if transition can be delayed by heating the surface of a vessel in water. Basically this was to be accomplished

by observing the reduction of frictional resistance at elevated surface temperatures. An appraisal of the practical utility of boundary layer control was the primary goal. For that reason a ship model towing tank procedure was employed. In selecting this method, it was anticipated that this thesis would be paralleled by an investigation of the effect in heated pipes, utilizing more easily controlled laboratory conditions. It was felt that the results of the towing tank experiment would serve to substantiate the results of the pipe experiments and to yield valuable information on the practical aspects of the theory since the towing tank conditions would more closely approximate the conditions that are likely to be encountered in an actual application.

II. PROCEDURE

MODEL DESIGN

Since the investigation was concerned with frictional resistance, the most advantageous model to use in the experiment would have been a completely submerged body or a friction plane. The M.I.T. Ship Model Towing Tank, however, is not at the present equipped with a towing carriage, which precluded the use of anything but a surface vessel which provided its own stability. Thus limited, a model was designed with the following aims and limitations:

1. Minimum residuary resistance was desired, since a large residuary resistance in comparison with the frictional resistance, would mask out the variations in the latter. Residuary resistance increases with increasing Froude number; hence the model should be run at as low a Froude number as possible.

2. In order to maximize the frictional resistance in comparison with the residuary resistance, a large wetted surface was desired.

3. Reference [12] indicates that laminar flow becomes unstable at a Reynolds number of approximately 4.5×10^5. For the purpose of our experiment, it was deemed advisable to ensure that the model could be run in or beyond the transition region. It was imperative that the model could be run at a Reynolds number to give

some turbulent flow in the unheated condition.

4. The beam had to be large enough to provide adequate stability and to allow enough space for the mounting of the towing bracket inside the model.

5. The other limitations of the M.I.T. Ship Model Towing Tank were that the maximum length could not exceed six feet and that, because of the inertia effect, the maximum displacement of the model could not exceed approximately thirty pounds.

The requirements of being able to run the model at a high Reynolds number to ensure turbulent flow and a low Froude number meant that the model should be as long as possible, or six feet in length. Further, the requirements of (1) also suggested that the bow and stern be adequately faired and that the beam be a minimum. The beam chosen, 5 inches, was the minimum which would meet the requirements of (4) above. The model would be run at the maximum displacement to give a large wetted surface.

The shape of the model hull as finally chosen is shown in the appendix (Fig. XII). Before proceeding with the fabrication of the metal model, a wooden model of the same shape was built and tested in the towing tank in order to see if the desired characteristics could be obtained. The curve of C_t versus Reynolds number obtained from the wooden model indicated that the turbulent region

could be reached, with a slight indication of the beginning of laminar flow at a Reynolds number of 7×10^5. This was considered satisfactory and the fabrication of the metal model was started at the Boston Naval Shipyard. The requirements of high thermal conductivity and corrosion resistance in water led to the selection of copper as the material used.

METHOD OF HEATING SURFACE OF MODEL

The optimum method of heating the surface of the model would have been a method for which no external connections to the model were required. This is basically because the equipment of the M.I.T. Ship Model Towing Tank does not include a towing carriage and any external connections are likely to introduce variable forces acting upon the model that would affect the reliability of the velocity measurements.

In an attempt to avoid external connections, the utilization of the heat of fusion of several available salts was considered. The weights required of several of the investigated salts indicated that their employment was reasonable. Elementary experimentation with these salts, however, showed the basic difficulty with the heat of fusion method to be progressive solidification. This process set up varying heat transfer barriers. To eliminate these barriers constant mixing would have been required and this was considered impractical.

Electrical heating was decided upon as the only practical alternative. This decision was made realizing that two important disadvantages of the method would have to be minimized. These disadvantages were the use of external connections and possible temperature variation along the hull.

It was decided to base the heat transfer rate calculations on the condition of laminar flow. The heating coil arrangement was, therefore, designed on the basis of the Pohlhausen equation for flat plates and laminar flow[11]. Number twenty AWG Nichrome V wire was selected because of its ability to withstand large currents and high temperatures. The resistance wires were spaced in the model so as to give constant surface temperatures.

Electrical insulation between the copper model and the heating wires proved an extremely difficult problem because of the high operating temperature of the wire and the need for a large heat conductivity through the electrical insulation. Experimentation with such materials as glass cloth and mica showed that they provided a heat transfer barrier that was formidable enough to either melt the wire or deteriorate the insulation. "Insa-Lute" (paste No. 1) was selected as the electrical insulation material because it showed itself to possess the required electrical and heat conduction properties. This material was employed by first coating the inner copper hull surface,

laying the wires, and then covering the wires with the same material. A final layer of heat insulating material was to be employed to reduce losses to the atmosphere. These loses were, however, deemed negligible in comparison with the heat transfer to the water and the final layer was not employed.

Connecting the electrical heating coils to the source of electrical power without introducing error in the towing tank measurements demanded much consideration. A survey of literature and several interviews provided no indication that electrical leads had ever been attached to a model in a towing tank that was not equipped with a towing carriage. The basic difficulty was, of course, that the electrical leads would tend to introduce unmeasurable forces thus preventing accurate determination of the effects of heating. It was decided that the wires should enter the model vertically with a minimum weight of wire being supported by the model. It was necessary to design a cart that could be moved along the side of the towing tank and parallel the motion of the model. In order to introduce the wires from above the model, a pole was attached to the cart and extended to a point directly above the model. The electrical leads were rigidly attached to the point on the pole above the model and slack in the leads was adjusted by trial and error. The external electrical leads were connected to the internal system at a three phase binding post that was

rigidly attached to the model in the bow section.

Surface temperatures were obtained from thermocouples that were soldered to the inner surface at three stations. These stations were placed respectively along the center-line of the bottom at 14.3 inches from the bow, 14.3 inches from the stern, and amidships. The thermocouple leads paralleled the external power leads from the cart to the model. The thermocouples were made of constantan and copper, and melting ice in water was used for the reference temperature.

The power supply available in the towing tank building dictated that three phase power be employed for heating the model, considering the estimated amounts of power required. The nine heating coils in the model were first connected in delta in groups of three coils. It was later decided to employ "Y" connection of the coils in order to lower the amount of line current for a given power input to the model. Voltage regulation was accomplished by employing a three phase variable transformer. The transformer, thermocouple potentiometer, and other electrical measuring equipment were located at the drive end of the towing tank.

DESCRIPTION OF TOWING TANK

The Ship Model Towing Tank at M.I.T. is described by M. A. Abkowitz [17] . Briefly, the tank is 108 feet long, 8 feet 7 inches wide, and 4 feet deep. The models

are accelerated by falling weights until the model resistance is equal to the towing force and travels thereafter at constant speed. The speed of the model is measured by means of electronic instrumentation at an idler pulley mounted at the far end of the tank. A black anodized sheet-aluminum disk mounted on the periphery of the idler pulley has 2,000 uniformly spaced radial slits. A light source and a phototube are mounted with the slotted disk between them, so that the phototube receives a light impulse every time a slit passes the center of the optical system. The output of the phototube is amplified and transmitted to an electronic counter located at the drive end of the tank. The precision of the instrumentation provides measurement of the towing force to within 0.0001 pounds, and measurement of the speed to 0.001 knots.

<u>TEST PROCEDURE</u>

The copper model was ballasted to a displacement of 31.27 pounds which gave a wetted surface of 4.81 square feet. A series of runs were then made with the power and thermocouple leads not connected to obtain the smooth hull characteristics. The turbulence was stimulated by a one-half inch wide sanded strip placed on the stem and a one inch wide sanded strip placed along the waterline on both sides extending aft twelve inches from the stem. Next, the wires were attached and runs were made with no

heat applied; then with heat applied. Three or four runs were made with the same towing force, both heated and unheated, and the recorded speeds were averaged.

The following observations were made from these early runs:

1) The expected point of inflection in the curve of C_t versus R_e at a Reynold's number of about $7x10^5$ indicating the start of transition to laminar flow either did not occur or was very slight if it did occur.

2) With the wires attached to the model, consistent results could not be obtained below a speed of 1.2 knots, indicating that the variable force introduced by the attachment of the leads was an appreciable percentage of the towing force at lower speeds.

The absence of transition was difficult to explain since the shape of the copper model was the same as the wooden model, and the metal model was considered to be smoother than the wooden model. One possible explanation was that the soldered joint connecting the bow to the parallel middle body offered a discontinuity to the flow and thus tripped the boundary layer into turbulent flow. To remove this contingency, melted solder was flowed on all portions of the hull in which there appeared to be irregularities. This solder was then scraped down until the hull was as free from defects as possible. Afterwards, the entire outer hull was buffed and polished to a mirror finish.

test applied; then with both applied. Three or four runs were made with the same towing force, each heated and unheated, and the resulting speeds were averaged.

The following observations were made from these towing runs:

1) The expected point of inflection in the curve of C_T versus R_N at a Reynold's number of about 7×10^5 indicating the start of transition to laminar flow either did not occur or was very slight if it did occur.

2) With the wires attached to the model, consistent results could not be obtained below a speed of 1.2 knots, indicating that the parasitic force introduced by the attachment of the leads was an appreciable percentage of the towing force at lower speeds.

The absence of transition was difficult to explain since the shape of the copper model was the same as the wooden model, and the metal model was considered to be smoother than the wooden model. One possible explanation was that the soldered joint connecting the bow to the parallel middle body offered a discontinuity to the flow and thus tripped the boundary layer into turbulent flow. To remove this contingency, melted solder was flowed on all portions of the hull in which there appeared to be irregularities. This solder was then scraped down until the hull was as free from defects as possible. Afterwards, the entire outer hull was buffed and polished to a mirror finish.

The friction of the entire towing assembly is of the order of 0.004 pounds and is calibrated as a function of the model speed. As the speed decreases, the towing friction as a percentage of the applied force on the weight pans increases. For example, at 0.25 knots the friction of the towing assembly was around 40 percent of the applied force for our model. Further, the friction has only been calibrated down to a speed of 0.4 knots and the friction for speed lower than 0.4 knots must be obtained by extrapolation of the friction curve. In view of the fact that the results become less reliable with decreasing speed, it was advantageous for our purposes to bring in cooler water so that a lower Reynold's number could be obtained for the same velocity of the model. By changing the water in the towing tank, the water temperature was reduced from 65 degrees F to 58 degrees F.

With these steps taken, a new curve of C_t versus R_e was obtained, as is discussed in section IV. Suffice to say here that the transition became more apparent at a Reynold's number of 7×10^5.

In order to obtain consistent results at lower speed with the wires attached, a slip-ring assembly was designed and built so that the power could be introduced into the model through carbon brushes. A drawing of the slip-ring assembly is shown in Figure X. With this device, the

The friction of the entire towing assembly is of the order of 2,000 pounds and is calibrated as a function of the towing speed. As the speed increases, the towing friction as a percentage of the applied force on the weight pans increases. For example, at 0.83 knots the friction of the towing assembly was around 60 percent of the applied force for our model. Further, the friction has only been calibrated down to a speed of 0.4 knots and the friction for speed lower than 0.4 knots must be obtained by extrapolation of the friction curve. In view of the fact that the results become less reliable with decreasing speed, it was advantageous for our purposes to bring in cooler water so that a lower Reynold's number could be obtained for the same velocity of the model. By changing the water in the towing tank, the water temperature was reduced from 65 degrees F to 38 degrees F.

With these steps taken, a new curve of C_T versus R_n was obtained, as is discussed in Section IV. Suffice to say here that the transition region more apparent at a Reynold's number of 7×10^5.

In order to obtain consistent results at lower speeds with the wires attached, a slip-ring assembly was designed and built so that the power could be introduced into the model through carbon brushes. A drawing of the slip-ring assembly is shown in Figure X. With this device, the

maximum force in the direction of travel that could be applied to the model by the power leads was the amount necessary to overcome the friction in the ball-bearings and slip-rings, providing the end limits of rotation of the assembly were not reached. With the wire attached to the slip-ring assembly, we were able to obtain consistent readings down to a model speed of 0.87 knots.

FIGURE I
COPPER MODEL & ORIGINAL WOODEN MODEL.

FIGURE II
INSIDE OF MODEL
WITH
TOWING BRACKET
&
SLIP RING ASSEMBLY
INSTALLED.

FIGURE III
MODEL DURING A TEST RUN.

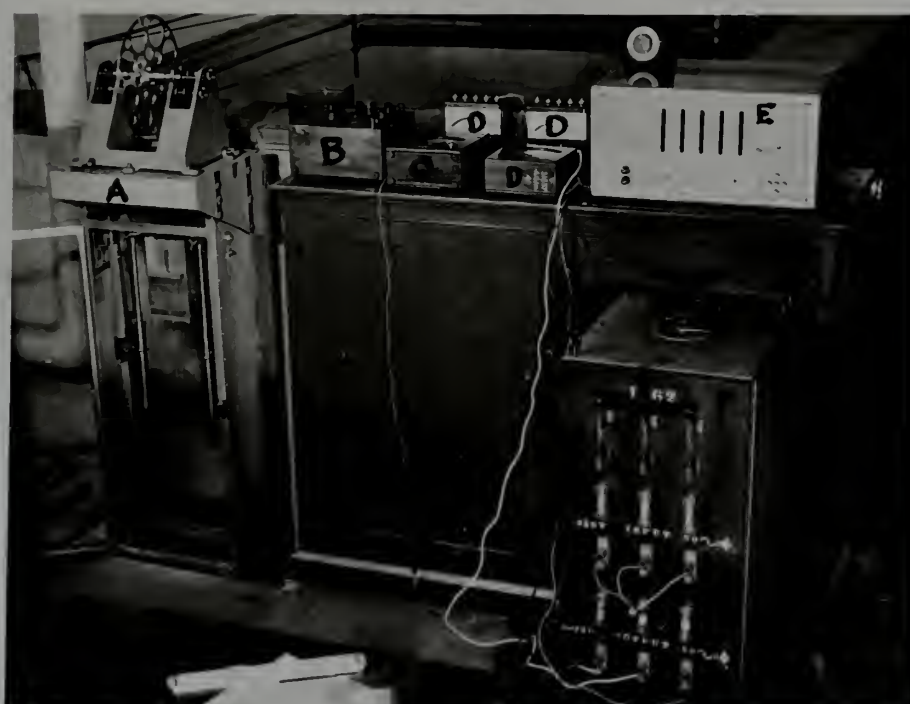


FIGURE IV
DRIVE END OF TOWING TANK.

A - DYNAMOMETER
B - THERMOCOUPLE POTENTIOMETER
C - VOLTMETER
D - AMMETERS
E - ELECTRONIC SPEED RECORDER
F - THREE PHASE VARIABLE TRANSFORMER

FIGURE V
INTERNAL VIEW OF MODEL
SHOWING ATTACHMENT OF EXTERNAL POWER LEADS.

FIGURE VI
HEATED MODEL RUN PROCEDURE.

A - CART & POLE B - THERMOCOUPLE REFERENCE TEMPERATURE SOURCE

III. RESULTS

Figure VII shows the relationship between C_t and R_e for the model when towed in accordance with the routine towing tank procedure at the indicated water temperatures. Points are included for the unheated model runs with the power leads attached.

Figure VIII shows the relationship between C_t and V for the following conditions:

1. Unheated model with smooth hull at indicated water temperatures.

2. Unheated model with smooth hull and power leads attached.

3. Heated model with smooth hull and power leads attached at various average surface temperatures of the hull. The surface temperatures, C_t and V for a given point are as indicated in Table I.

Table I gives the experimental results obtained at the various points where the effect of heating was investigated. T_h is the calculated boundary layer temperature that would be required to produce the same effect as that which was experimentally determined. It is computed on the assumption that the flow past the hull is turbulent, and on the basis of Schoenherr's friction formulation. Details of this computation are given in the appendix.

Figure IX shows the relationship between power input per degree Fahrenheit temperature difference (between

VII. RESULTS

Figure VII shows the relationship between C_t and R_n for the model when towed in accordance with the routine towing tank procedure at the indicated water temperature. Points are included for the unheated model runs with the power leads attached.

Figure VIII shows the relationship between C_t and V for the following conditions:

1. Unheated model with smooth hull at indicated water temperature.

2. Unheated model with smooth hull and power leads attached.

3. Heated model with smooth hull and power leads attached at various average surface temperatures of the hull. The surface temperatures, C_t and V for a given point are as indicated in Table I.

Table I gives the experimental results obtained at the various points where the effect of heating was investigated. T_B is the calculated boundary layer temperature that would be required to produce the same effect as that which was experimentally determined. It is computed on the assumption that the flow past the hull is turbulent, and on the basis of Schoenherr's friction formulation. Details of this computation are given in the appendix.

Figure IX shows the relationship between power input per degree Fahrenheit temperature difference (between

surface and water temperature) and velocity. It gives a comparison of the experimentally determined values on the average hull surface temperature and the theoretical relationship based upon Pohlhausen's theoretical equation for flat plates and laminar flow[11].

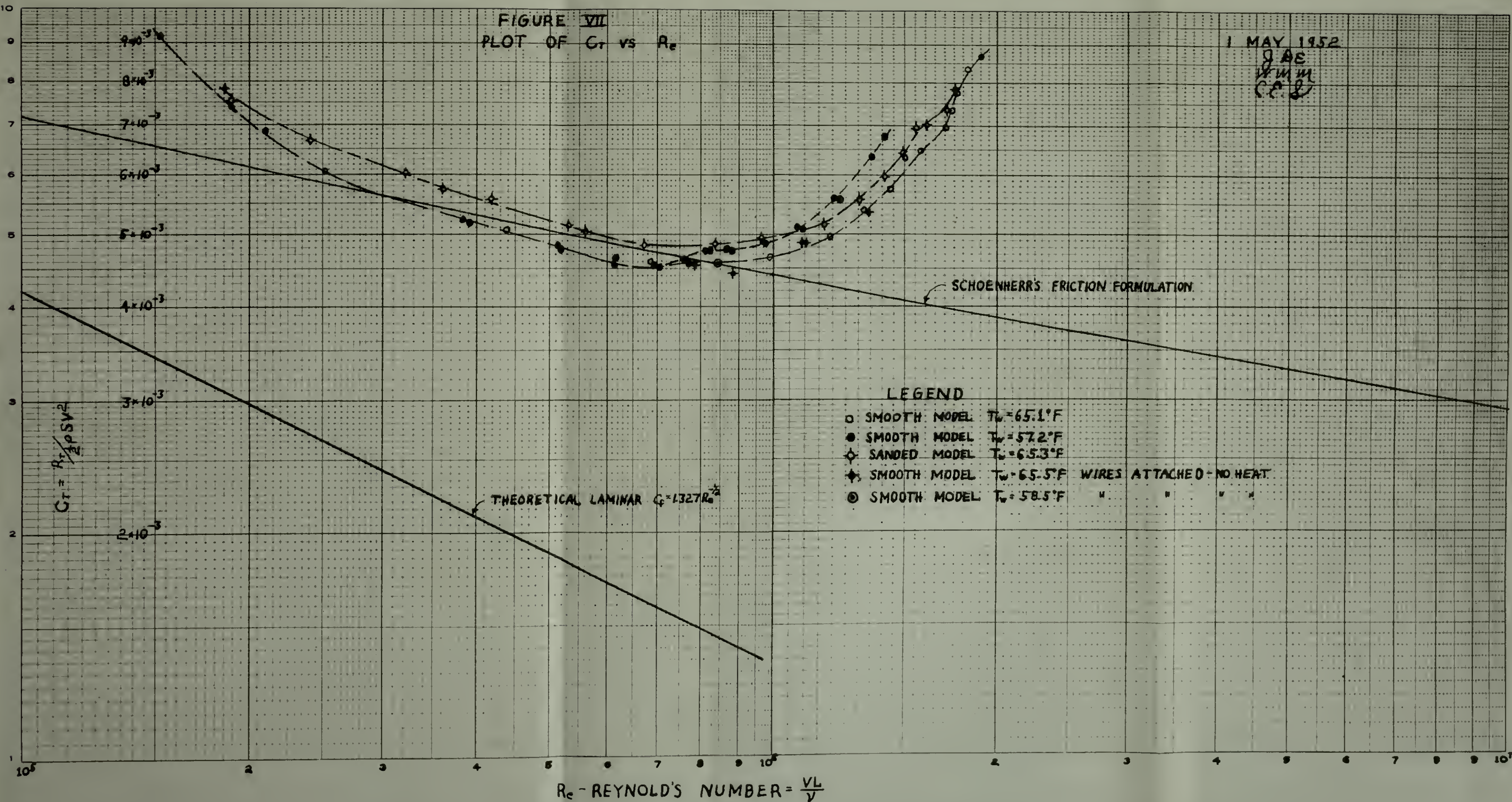


FIGURE VII
PLOT OF C_T vs R_e
1 MAY 1952
J.A.E.
W.M.M.
C.E.G.
9×10^{-3}
8×10^{-3}
7×10^{-3}
6×10^{-3}
5×10^{-3}
4×10^{-3}
3×10^{-3}
2×10^{-3}
$C_T = \frac{R_T}{\frac{1}{2}\rho S V^2}$
SCHOENHERR'S FRICTION FORMULATION
THEORETICAL LAMINAR $C_f = 1.327 R_e^{-\frac{1}{2}}$
LEGEND
SMOOTH MODEL $T_w = 65.1°F$
SMOOTH MODEL $T_w = 57.2°F$
SANDED MODEL $T_w = 65.3°F$
SMOOTH MODEL $T_w = 65.5°F$ WIRES ATTACHED - NO HEAT
SMOOTH MODEL $T_w = 58.5°F$ " " " "
10^5
10^6
10^7
R_e - REYNOLD'S NUMBER = $\frac{VL}{\nu}$

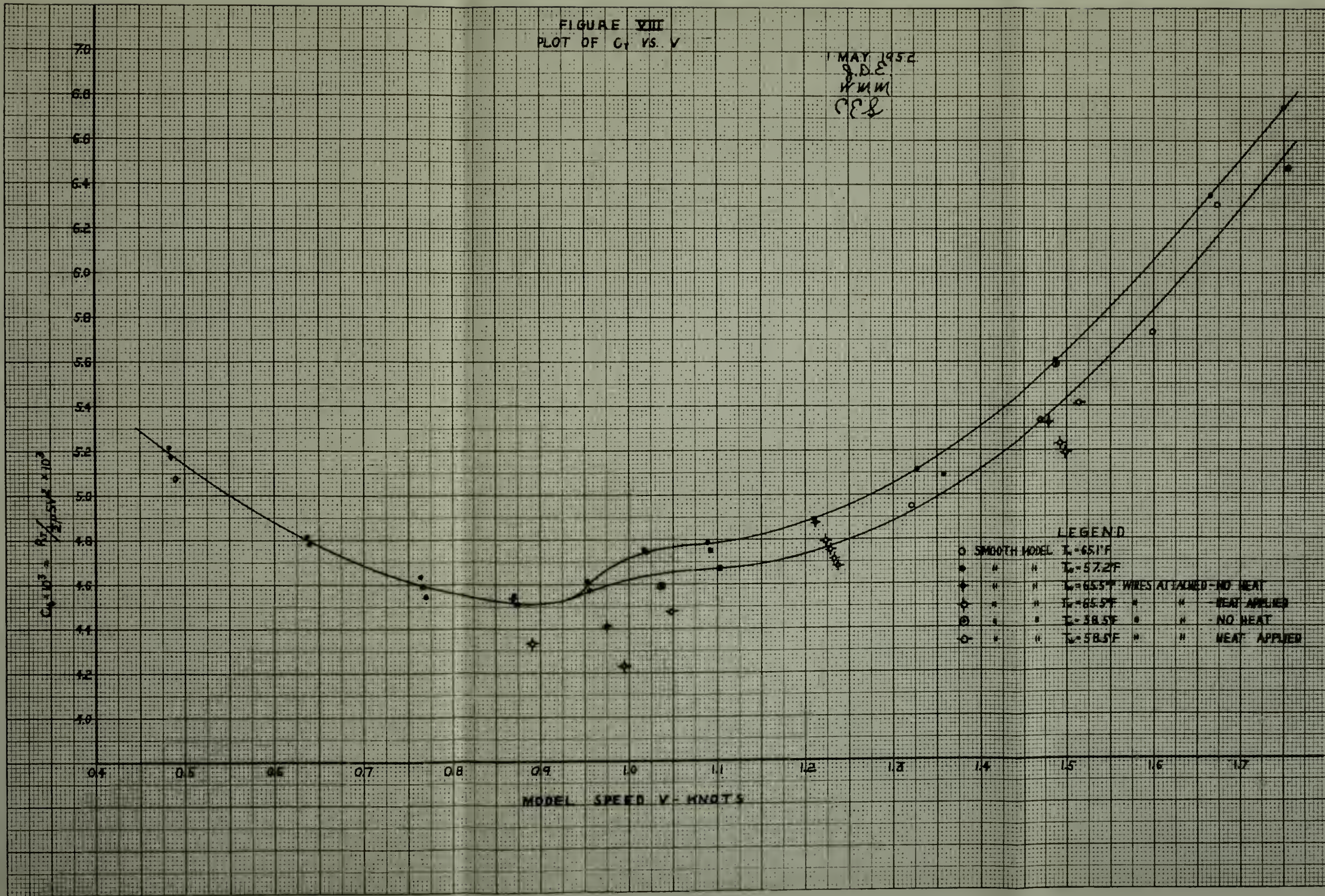


FIGURE VIII
PLOT OF C_T VS. V
1 MAY 1952
J.D.E.
H.M.M.
C.E.L.
7.0
6.8
6.6
6.4
6.2
6.0
5.8
5.6
5.4
5.2
5.0
4.8
4.6
4.4
4.2
4.0
$C_T \times 10^3 = R_T / \tfrac{1}{2}\rho S V^2 \times 10^3$
0.4
0.5
0.6
0.7
0.8
0.9
1.0
1.1
1.2
1.3
1.4
1.5
1.6
1.7
MODEL SPEED V - KNOTS
LEGEND
SMOOTH MODEL T_w = 65.1°F
" " T_w = 57.2°F
" " T_w = 65.5°F WIRES ATTACHED - NO HEAT
" " T_w = 65.5°F " " - HEAT APPLIED
" " T_w = 58.5°F " " - NO HEAT
" " T_w = 58.5°F " " HEAT APPLIED

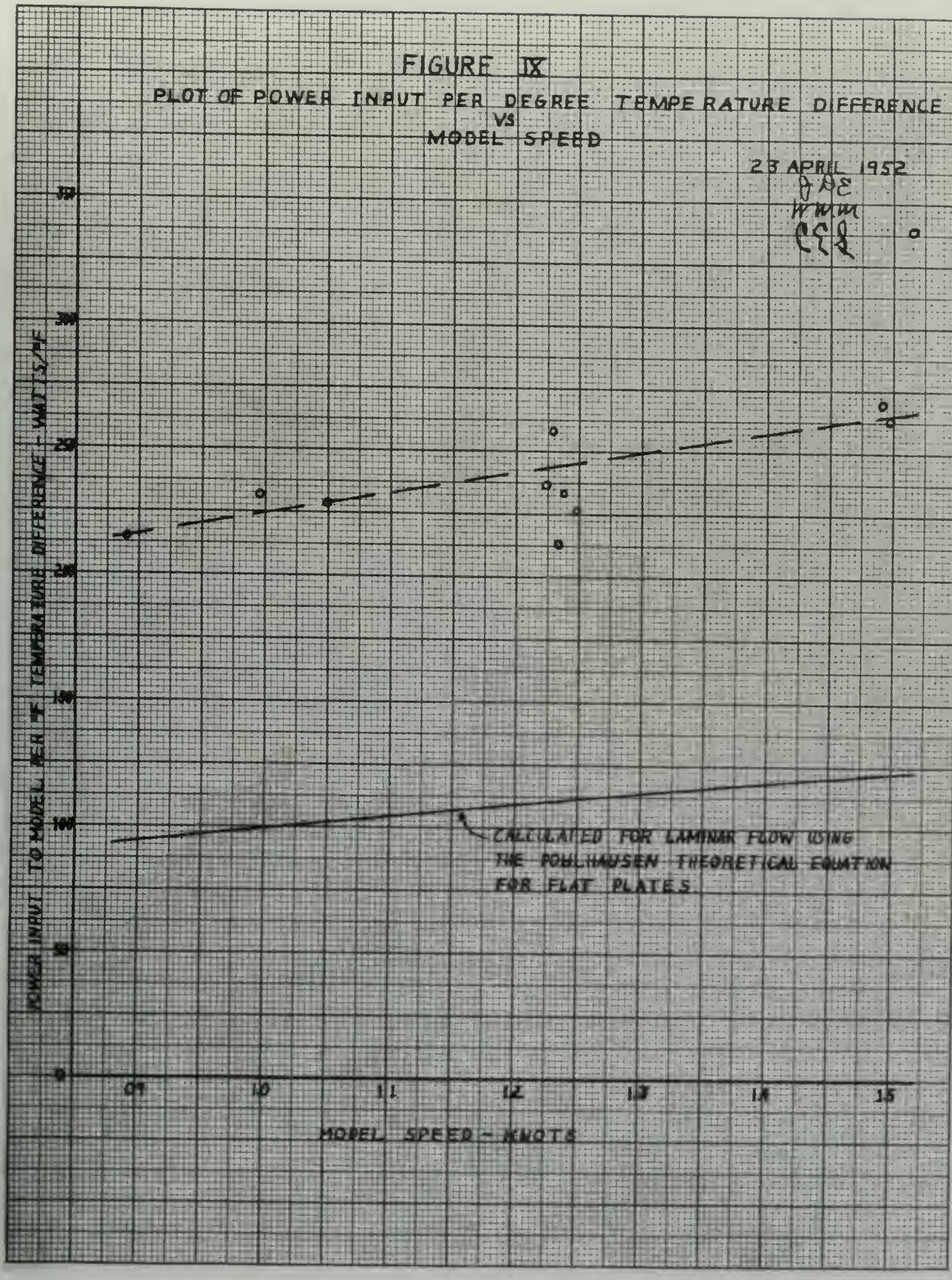


FIGURE IX
PLOT OF POWER INPUT PER DEGREE TEMPERATURE DIFFERENCE
VS
MODEL SPEED
23 APRIL 1952
POWER INPUT TO MODEL PER °F TEMPERATURE DIFFERENCE – WATTS/°F
350
300
250
200
150
100
50
0
0.9
1.0
1.1
1.2
1.3
1.4
1.5
MODEL SPEED – KNOTS
CALCULATED FOR LAMINAR FLOW USING THE POHLHAUSEN THEORETICAL EQUATION FOR FLAT PLATES.

IV. DISCUSSION OF RESULTS

The primary purpose of obtaining the information shown in Figure VII for the smooth and sanded hull was to establish the Reynolds number at which transition from laminar to turbulent flow occurred. Once this region was firmly established, the model could be towed at higher Reynolds numbers with the surface heated, and the effect of the heating on transition noted.

The region of transition was not apparent from the initial tests with a water temperature of about 65°F. In general, past experience with models of similar form characteristics would lead one to expect a pronounced transition region showing distinctively laminar flow at the lower Reynolds number [16].

The curve of C_t for a water temperature of about 58°F, as was expected, separated from the curve obtained at about 65°F. in the region where the residuary resistance became noticeable. This is, of course, explained by the fact that at the same Reynolds number, the two runs at different temperatures have different Froude numbers, hence different residuary resistances.

This curve of C_t for a water temperature of about 58°F. shows a distinct inflection in the region of Reynolds numbers of $7x10^5$ to $8.5x10^5$. However at still lower Reynolds numbers the curve of C_t rises above the Schoenherr friction curve

IV. DISCUSSION OF RESULTS

The primary purpose of obtaining the information shown in Figure VII for the smooth and rough hull was to establish the Reynolds number at which transition from laminar to turbulent flow occurred. Once this region was firmly established, the model could be towed at higher Reynolds numbers with the surface heated, and the effect of the heating on transition noted.

The region of transition was not apparent from the initial tests with a water temperature of about 65°F. In general, past experience with models of similar form characteristics would lead one to expect a pronounced transition region showing distinctively laminar flow at the lower Reynolds number [15].

The curve of C_T for a water temperature of about 90°F, as was expected, separated from the curve obtained at about 65°F. in the region where the residuary resistance became noticeable. This is, of course, explained by the fact that at the same Reynolds number, the two runs at different temperatures have different Froude numbers, hence different residuary resistances.

This curve of C_T for a water temperature of about 90°F. shows a distinct inflection in the region of Reynolds numbers of 7×10^5 to 8.5×10^5. However at still lower Reynolds numbers the curve of C_T rises above the Schoenherr friction curve

instead of tending downward toward the Blasius curve as would be expected in the case of laminar flow. This condition has not been satisfactorily explained. If one assumes that below a Reynolds number of $7x10^5$ the flow is essentially laminar, the high values of C_t could possibly be caused by a sharp increase in pressure drag due to laminar separation at the converging section of the stern. Attempts to prevent separation by sanding the stern section were unsuccessful.

Another factor that casts some doubt on the accuracy of the results at low Reynolds numbers is the possibility that the friction calibration of the towing equipment at speeds below 0.4 knots may be in error by a considerable amount since it is obtained by extrapolation of the measured friction for speeds of 0.4 knots and above. At the lower speeds, friction is an appreciable percentage of the applied force.

In spite of the foregoing, there are indications that the flow was essentially turbulent throughout the range of Reynolds numbers. These indications are the lack of a definite transition region and the fact that the curve of C_t remains in close proximity to the Schoenherr turbulent curve even at low Reynolds numbers. It is possible that the apparent inflection in the curve of C_t at Reynolds numbers of $7x10^5$ to $8.5x10^5$ is due to

a sharp increase in residuary resistance due to wave formation.

Further substantiation of the absence of laminar flow during the heated runs is found in Figure IX. The heat transfer rate through a turbulent boundary layer has been found to be much greater for the same temperature difference and velocity than that through a laminar boundary layer [10] . As shown in Figure IX, our calculated heat transfer rate to the water (neglecting the small amount of heat being transferred to the air) is more than twice the rate that we would expect if the boundary layer was completely stabilized to laminar flow.

The results of heated runs are plotted in Figure VIII. A small but significant change in C_t resulted when the model surface was heated.

A more significant indication of the effect of heating the surface of the model is shown in Table I where T_h is compared to the average surface temperature (T_{as}). Since all but four of the runs show values of T_h which are less than the average surface temperature, it appears that the change in C_t caused by heating was due only to the viscosity change in the boundary layer and not to any delay in transition. For the four runs in which T_h exceeded the average surface temperature, the average value of (T_h - T_{as}) was 3.7 degrees F., while

the maximum value of $(T_h - T_{as})$ was only 6.8 degrees F. Further, these four runs, in general, occurred at the higher speeds where the change of C_r with velocity was much larger and the evaluation of the change more subject to error. Since the determination of T_h depended on the accurate evaluation of the change in C_r, the possible error in T_h increases with an increase in speed. This fact, coupled with the expectation that a much greater reduction in resistance would have occurred if the transition to laminar flow had been delayed, casts doubt on the validity of concluding on the basis of these four runs, that the nature of flow past the model was changed in any way.

No definite conclusions can be drawn as to whether or not any stabilization of flow took place through heating until a more accurate determination of the character of flow past the model and the temperature distribution in the model hull is accomplished. But the weight of evidence from our limited data tends to support the conclusion that the change brought about by heating was caused only by the variation in viscosity.

By way of summary, the identity of the type of flow around the tested model was not firmly established. The effect of heating that was observed by this method of investigation is most likely chargeable to the

viscosity effect. The indications that the transition from laminar to turbulent flow was delayed by heating were not strong enough to substantiate or disprove the effect under investigation.

It is suggested that this investigation be continued by another method of experimentation. The applicable methods of experimentation include flow through pipes, circulating water channels, propeller tunnels, and towing tanks that are equipped with a towing carriage. These methods would provide a larger range of Reynolds numbers than the range that was available for this initial investigation. This investigation was limited by large form drag at the higher Reynolds numbers and by power lead errors at the lower Reynolds numbers. It is felt that any of the above listed methods would lend themselves to providing power in large quantities to heat the surfaces without introducing errors in measured forces and speeds. It is also recommended that facilities for study of the character of flow be incorporated in any of the above types of investigation of this subject.

V. CONCLUSIONS

1. The results obtained from heating the surface of the towed model are not sufficiently conclusive to substantiate or disprove the effect under investigation.

2. The need for external power leads makes the use of a towing tank which does not include a towing carriage an unsatisfactory method for investigating the delay of boundary layer transition along a heated surface.

VI. RECOMMENDATIONS

1. The investigation should be continued by one of the following methods of experimentation:

(a) Flow through pipes

(b) Circulating water channels

(c) Propeller tunnels

(d) Towing tanks that are equipped with a towing carriage.

2. Facilities for the study of the character of flow should be included in any future investigation.

APPENDIX

APPENDIX A

DETAILS OF PROCEDURE

HEATING COIL DESIGN

The heating coil arrangement was designed on the basis of the Pohlhausen equation for laminar flow past flat plates [11] which provided a relationship between longitudinal position along the hull and theoretical heat transfer rate per unit area. This enabled the determination of longitudinal spacing of the wires which would theoretically give a constant hull surface temperature.

The design of the heating coil arrangement consisted of the following steps:

1. #2 0 AWG Nichrome V wire (B-82, Ni-Cr) was selected for heating based on its ability to withstand large currents and high temperatures.

2. Starting from the bow,and working aft, a current was selected for each run of wire such that the resultant spacing of the heating wire would be neither too close and cause installation difficulties nor too far apart and exaggerate the heating discontinuity.

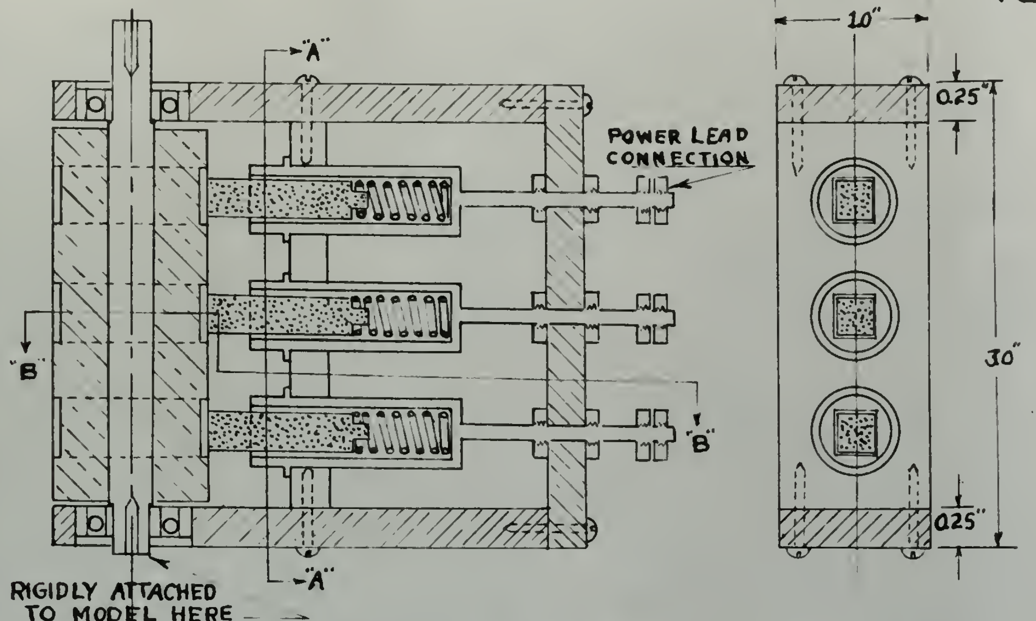
3. With the current selected, and assuming a voltage of 110 volts, the length of wire for each particular run was calculated. Also the theoretical heat output per inch of wire was determined.

4. With the above information available, the spacing of each pass of wire was determined. By this procedure, it was determined that nine separate heating wires were required with a total length of 120.67 ft. Figure XI shows the final arrangement of the wires.

FIGURE X

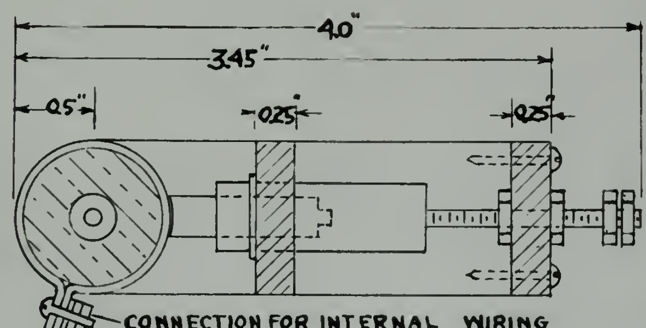
THREE PHASE SLIP-RING ASSEMBLY

28 APRIL 1952



SECTIONAL ELEVATION

SECTION "A-A"



SECTION "B B"

LEGEND

- COMPOSITION (INSULATOR)
- BAKELITE SPINDLE
- GRAPHITE BRUSHES

BRASS BRUSH HOLDERS AND COPPER SLIP RINGS NOT CROSS-HATCHED.

SCALE: FULL SIZE

THREE PHASE SLIP RING ASSEMBLY


#III WIRE - L = 11'-6.8"

WIRE No.
121
122
123
124
125
126
127
128
129
131
132
133
134
135
140
141
142
143
144
145
146
5.49
#XI WIRE - L = 11'-3"

FIGURE XI

33

PLAN AND ELEVATION

FIGURE XI

APPENDIX B

SUMMARY OF DATA AND CALCULATIONS

T_h Calculations

1. For a given applied force the velocity of the heated and the unheated model were observed. By standard towing tank calculations the total resistance coefficients, based upon the measured water temperature, were computed.

2. The respective residual resistance coefficients were read off of the previously drawn curve of residual resistance coefficient versus velocity curve. (Figure XII).

3. Having computed the Reynolds number of the unheated model run, the frictional resistance coefficient was obtained from tables of the Schoenherr Frictional Resistance Coefficients versus Reynolds Numbers. [13]

4. Realizing that the total resistance coefficient changes as the inverse of water density, it was assumed that the density remained constant over the range in water temperatures between the heated and unheated runs. Therefore it was noted that the difference in total resistance coefficients that were calculated as indicated above were equal to the sum of the differences of the residual resistance coefficients and the frictional resistance coefficients.

5. The only unknown in the relationship stated in (4.) was the frictional resistance coefficient for the

heated model run. It is obtained from that relationship.

6. Entering the tables of the Schoenherr Frictional Resistance Coefficients versus Reynolds Numbers with the determined value of frictional resistance coefficient for the heated model run, the corresponding Reynolds number was obtained.

7. Knowing all the terms in the Reynolds number except the kinematic viscosity of the water, allowed determination of that value.

8. Entering the table of kinematic viscosity versus water temperature [13] yielded the desired value of T_h.

<u>Actual and Theoretical Power Input to the Model per Degree of Temperature Difference</u>

1. The actual power input per degree of temperature difference was obtained by taking the calculated values of power input for each heated run as listed in column 5 of Table I and dividing these values by the difference between the average surface temperature as obtained by thermocouple readings and the water temperature as observed by thermometer.

2. Substituting the properties of water at 65.4°F in the Pohlhausen theoretical equation for laminar flow past flat plates [11] and combining the results with the wetted surface of 4.81 square feet, provided the theoretical power input to the model per degree of

temperature difference at any selected value of model velocity.

3. The two relationships that were calculated as indicated above are plotted for comparison in Figure IX.

Power Input Calculations

1. With a Simpson OHM meter the following resistances were obtained.

COIL	RESISTANCE (Ohms)
1	6.30
2	7.75
3	7.85
4	8.80
5	9.00
6	7.50
7	9.00
8	11.10
9	12.40

2. The coils were paralleled in the following groupings:

GROUP	COILS INCLUDED
I	2, 3, 8
II	1, 7, 9
III	4, 5, 6

3. The equivalent resistance of each of the paralleled groups was calculated.

4. For determination of the power input for "Y" connection of the groups, the average measured line current squared was multiplied by the sum of the group equivalent resistances.

5. For determination of the power input for "Delta" connection of the groups, the average measured line current squared was divided by the square root of three and multiplied by the sum of the group equivalent resistances.

6. The result of (4.) and (5.) above were checked with calculations using the measured line voltage and calculated line drop.

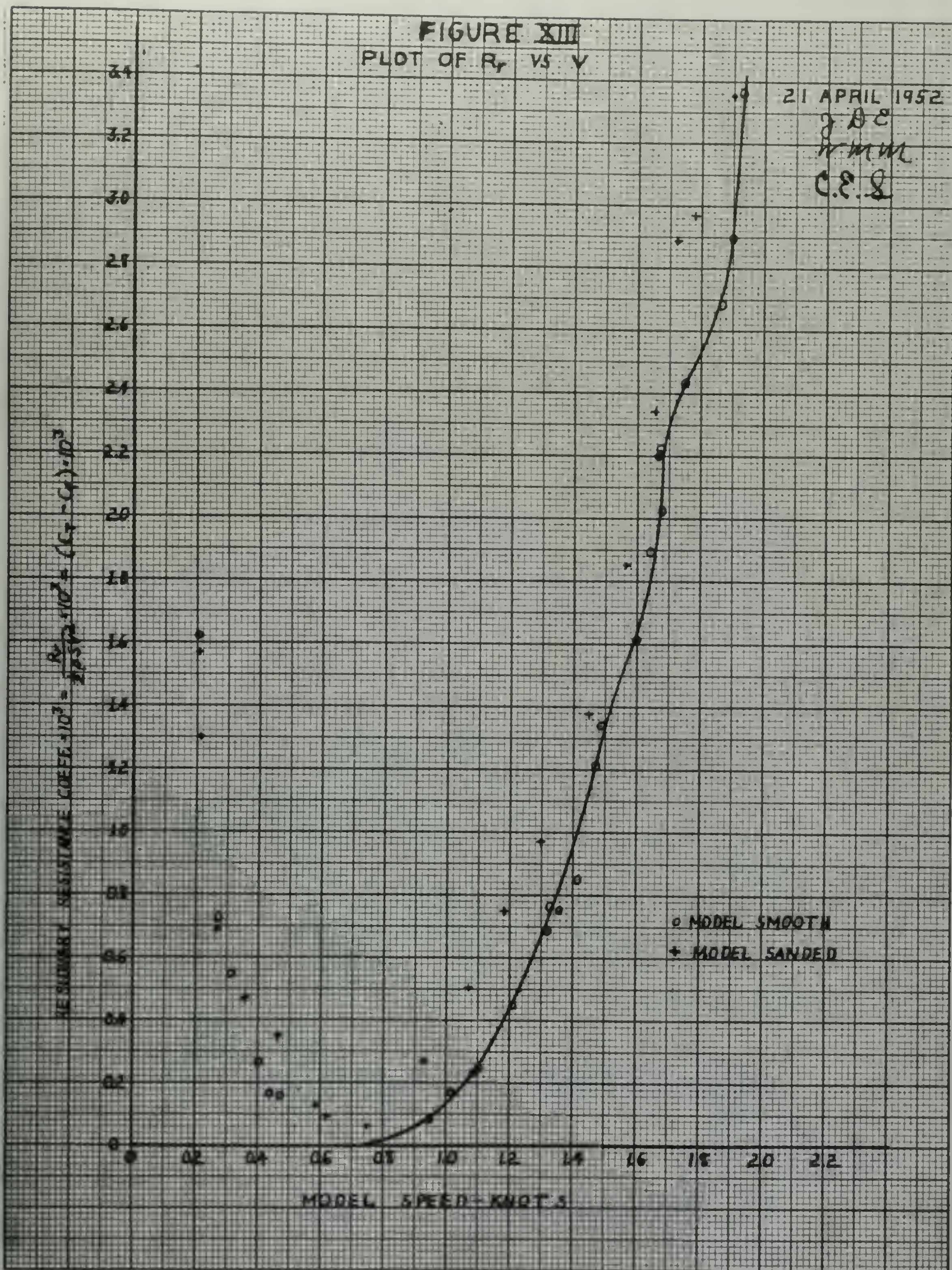


FIGURE XIII
PLOT OF R_r VS V

APPENDIX C

ORIGINAL DATA

TABLE I

Data From Heated and Unheated Runs with Electrical Leads Attached

Applied Force (lbs.)	Average Velocity (Kts.)	Max. Velocity Variation from Average (Kts.)	Water Temp. (°F)	Power Input (KW.)	Dynamometer Friction (lbs.)	R_t (lbs)	C_t x10^3	R_e x10^{-5}	Surface Temp (°F) Bow	Mid	Stern	Average T_{as}	T_h
.05	.869	.005	65.4	0	.00440	.04560	4.544	7.866					
.05	.890	.004	65.4	6.69	.00443	.04557	4.330	8.056	97.4	109.4	82.7	96.5	88.4
.06	.973	.001	65.7	0	.00453	.05547	4.410	8.820					
.06	.994	.002	65.7	6.69	.00455	.05545	4.224	9.011	96.3	105.2	82.1	94.5	76.1
.07	1.036	.001	58.2	0	.00460	.06540	4.583	8.458					
.07	1.048	.001	58.2	6.69	.00452	.06538	4.477	8.556	88.0	97.7	76.4	87.4	67.5
.10	1.213	.003	65.8	0	.00481	.09519	4.869	11.011					
.10	1.223	.004	65.8	2.56	.00483	.09517	4.789	11.102	76.5	80.9	72.5	76.7	75.0
.10	1.237	.003	65.8	4.55	.00485	.09515	4.680	11.229	85.3	94.2	76.5	85.3	89.3
.10	1.212	.007	65.2	0	.00482	.09518	4.877	10.912					
.10	1.228	.002	65.2	6.69	.00484	.09516	4.748	11.056	92.8	102.5	78.1	91.1	80.7
.10	1.233	.001	65.2	7.69	.00485	.09515	4.710	11.101	103.0	112.0	89.0	101.3	84.9
.16	1.480	.002	65.8	0	.00515	.15485	5.321	13.435					
.16	1.492	.001	65.8	2.56	.00517	.15483	5.235	13.544	75.1	77.8	72.9	75.3	78.5
.16	1.498	.003	65.8	4.55	.00517	.15483	5.193	13.598	83.1	92.0	74.2	83.1	84.0
.17	1.489	.001	58.9	0	.00516	.16483	5.592	12.265					
.17	1.515	.005	58.9	6.69	.00519	.16481	5.481	12.479	76.7	85.5	73.6	78.6	85.4

TABLE II
Data from Smooth Model Runs Without Electrical Leads

18 & 19 March 1952

Run No.	Water Temp. T_w(^oF)	Speed V(Kts.)	Applied Force (lbs.)	Dynamometer Friction (lbs.)	Net Resistance R_t (lbs.)	R_t SV^2 x10^3 C_tx10^3	Reynolds No. x10^{-5} Re x 10^{-5}
1.	65.1	0.280	.01	.00368	.00632	6.066	2.518
2.	65.1	0.488	.02	.00394	.01606	5.076	4.388
3.	65.1	0.765	.04	.00428	.03572	4.593	6.878
4.	65.1	1.102	.08	.00470	.07530	4.666	9.908
5.	65.1	1.322	.12	.00496	.11504	4.954	11.886
6.	65.1	1.470	.16	.00515	.15485	5.393	13.217
7.	65.1	1.599	.20	.00530	.19470	5.731	14.377
8.	65.1	1.673	.24	.00540	.23460	6.308	15.042
9.	65.1	1.753	.28	.00550	.26450	6.477	15.762
10.	65.1	1.899	.34	.00565	.33435	6.977	17.074
11.	65.1	1.933	.37	.00570	.36430	7.337	17.380
12.	65.3	1.960	.40	.00575	.39425	7.724	17.670
13.	65.3	2.024	.46	.00582	.45418	8.335	18.256
14.	65.3	2.111	.52	.00592	.51408	8.682	19.031

TABLE III

Data From Smooth Model Runs Without Electrical Leads 10 & 11 April, 1952

Run No.	Water Temp. T_w (°F)	Speed V(KTS)	Applied Force (Lbs.)	Dynamometer Friction (lbs.)	Net Resistance R_t(lbs.)	$\frac{R_t}{\frac{\rho}{2} SV^2} \times 10^3$ $C_t \times 10^3$	Reynolds No. $\times 10^{-5}$ $R_e \times 10^{-5}$
1.	56.8	0.191	.008	.00355	.00445	9.173	1.525
2.	57.6	0.234	.009	.00362	.00538	7.389	1.891
3.	56.8	0.263	.010	.00367	.00633	6.852	2.100
4.	56.8	0.481	.020	.00395	.01605	5.217	3.841
5.	57.6	0.483	.020	.00394	.01606	5.177	3.903
6.	56.8	0.638	.030	.00412	.02588	4.781	5.095
7.	57.6	0.636	.030	.00412	.02588	4.812	5.140
8.	56.8	0.769	.040	.00428	.03572	4.542	6.142
9.	57.6	0.763	.040	.00426	.03574	4.639	6.166
10.	56.8	0.869	.050	.00440	.04560	4.544	6.940
11.	57.6	0.873	.050	.00441	.04559	4.501	7.055
12.	56.8	0.951	.060	.00450	.05550	4.615	7.595
13.	57.6	0.955	.060	.00450	.05550	4.576	7.718
14.	56.8	1.018	.070	.00458	.06542	4.747	8.130
15.	57.6	1.017	.070	.00458	.06542	4.757	8.219
16.	56.8	1.088	.080	.00467	.07533	4.766	8.689
17.	57.6	1.092	.080	.00469	.07531	4.750	8.825
18.	56.9	1.210	.100	.00480	.09520	4.890	9.707
19.	57.6	1.211	.100	.00482	.09518	4.881	9.786
20.	57.6	1.328	.125	.00497	.12003	5.118	10.732
21.	56.9	1.359	.130	.00500	.12500	5.090	10.902
22.	57.6	1.488	.170	.00516	.16484	5.599	12.025
23.	57.9	1.666	.240	.00540	.23460	6.357	13.523
24.	57.9	1.748	.280	.00550	.27450	6.756	14.188

TABLE IV

Data From Sanded Model Runs without Electrical Leads Attached

24 March 1952

Run No.	Water Temp. TW(^{o}F)	Speed V(Kts.)	Applied Force (lbs.)	Dynamometer Friction (lbs.)	Net Resistance R_t(lbs.)	$\frac{R_t}{SV^2}$ x10^3 C_tx10^3	Reynolds No.x10^5 Re x 10^{-5}
1	65.3	0.210	.008	.00360	.00440	7.509	1.893
2	65.3	0.267	.010	.00367	.00633	6.683	2.407
3	65.3	0.358	.014	.00377	.01023	6.007	3.227
4	65.3	0.400	.016	.00382	.01218	5.729	3.606
5	65.3	0.465	.020	.00393	.01607	5.593	4.192
6.	65.3	0.588	.027	.00405	.02355	5.126	5.301
7.	65.3	0.622	.030	.00410	.02590	5.038	5.607
8.	65.3	0.747	.040	.00425	.03575	4.822	6.734
9.	65.3	0.930	.060	.00448	.05552	4.831	8.384
10.	65.3	1.071	.080	.00465	.07535	4.944	9.655
11.	65.3	1.184	.100	.00480	.09520	5.111	10.674
12.	65.3	1.297	.120	.00492	.11508	5.148	11.693
13.	65.3	1.447	.160	.00510	.15490	5.568	13.045
14.	65.3	1.566	.200	.00525	.19475	5.977	14.118
15.	65.3	1.658	.240	.00537	.23463	6.424	14.947
16.	65.3	1.726	.280	.00546	.27454	6.936	15.560
17.	65.3	1.779	.300	.00551	.29449	7.003	16.038
18.	65.3	1.907	.360	.00568	.35432	7.333	17.192
19.	65.3	1.949	.400	.00574	.39426	7.811	17.571

APPENDIX D

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